



CFD simulation of thermoacoustic cooling

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ABSTRACT

In the field of thermoacoustic energy conversion, the application of numerical analysis techniques, specifically computational fluid dynamics (CFD) simulations, have gained ground in recent years. Previous efforts have focused on single thermoacoustic couples that were subjected to the thermoacoustic effect through an oscillatory boundary condition. CFD simulations of an entire thermoacoustic device are computationally expensive and few examples exist. The present work presents an extension of a simulation of a whole thermoacoustic engine that also includes a refrigeration stack. Through interaction of thermally generated sound waves, cooling of the working gas in this stack is demonstrated.

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1. Introduction

The basic thermodynamic cycle in thermoacoustic devices is the Stirling cycle, which was developed in 1816 [1]. The original mechanical Stirling engine utilized two pistons and a regenerative heat exchanger [2]. Over the course of one cycle, the working gas is compressed, and it then transfers thermal energy to the heat sink, thus maintaining a constant temperature. Next, the gas is heated at constant volume by the regenerator and then is heated further at the heat source. This heat supply occurs while the gas is allowed to expand and drive the power piston, again at constant temperature. After expansion, the gas is displaced to the heat sink, while cooling at constant volume by depositing heat to the regenerator, which stores heat between cycle segments [2]. The first application of this cycle as a thermoacoustic technology occurred when Ceperley recognized that sound waves could replace pistons for gas compression and displacement [3].

1.1. Thermoacoustic engines

The simplest thermoacoustic engines (TAEs) operate with a standing acoustic wave in a resonance tube. This wave causes pressure variations as well as gas displacement. In a quarter wavelength resonator comprised of a tube with one open end and one closed end, the pressure anti-node of the wave is located at the closed end, and the velocity anti-node is located at the open end. When a gas is subject to such a standing wave and it interacts with a solid wall that is subject to a (externally imposed) temperature

gradient, pressure disturbances can be intensified to reach strong amplitudes. In practice, this interaction occurs within a regenerative unit commonly referred to as the “stack” in standing wave devices. Ceramic monolith structures are often used as stacks because they offer a large surface-to-volume ratio and exhibit desirable hydraulic performance. In order for amplification to occur, this temperature gradient must be larger than the critical temperature gradient, which is related to the temperature gradient that the gas would experience if it were under the influence of a sound wave in adiabatic conditions. The expression for this critical temperature gradient

$$\nabla T_{crit} = \frac{\omega p_1^s}{\rho_m c_p u_1^s} \quad (1)$$

was derived by Swift [4]. It depends on the operating frequency ω , the first order pressure and velocity in the standing wave p_1^s and u_1^s , as well as the mean gas density ρ_m and specific heat c_p .

1.2. Thermoacoustic refrigerators without moving parts

As stated above, an imposed temperature gradient along a porous medium such as a ceramic monolith placed in a resonance tube can produce strong pressure oscillations through the thermoacoustic effect. Again, it is based on the transfer of heat between the working gas and the solid walls at two different temperature levels and pressures. If we allow strong sound waves to interact with a porous medium, this cycle is reversed; the pressure amplitude is attenuated and a temperature gradient can be created along this porous medium. This phenomenon forms the basis of thermoacoustic refrigeration. Simple systems utilize mechanical drivers (such as speakers) to

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Nomenclature

c	specific heat
C_{fd}, C_{sf}	correlation constants
d	diameter
f	frequency
p	pressure
Q	heat flux
Re	Reynolds number
T	temperature
u	velocity
x	position
x^*	position relative to left stack edge

Greek symbols

λ	wavelength
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ω	angular frequency
ρ	density

Subscripts

1	first order
crit	critical
m	mean, time averaged
p	constant pressure
w	wire

Superscripts

s	standing wave
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create the pressure oscillations. However, a TAE can also be utilized to drive this thermoacoustic refrigerator (TAR). Such TARs are a reality today; however they are limited to few, specific uses, such as gas liquefaction. There are several examples of advanced TARs used for these specialized applications. Radebaugh at the National Institute of Standards and Technology (NIST) built a TAR with 5 W of cooling power at 120 K and a low temperature at no load of 90 K [5,6]. The main benefit is the lack of moving parts (such as seals), thus reducing maintenance costs. It is noteworthy that TARs can achieve these low temperatures in a single stage, whereas conventional vapor compression refrigerators (VCRs) can only achieve ≈ 230 K in a single stage [6]. Cryogenic cooling is not the only application for TARs. A practical example of this design was given by Poese with a freezer for ice cream storage. This small scale chiller featured an annular space around the regenerative unit to achieve traveling wave phasing [7]. The temperature gradient achievable by thermoacoustic cooling is limited by the critical temperature gradient.

2. Simulation of the thermoacoustic effect

Baseline numerical simulations of the thermoacoustic effect have been demonstrated several times. Early modeling efforts done by Cao et al. considered a reduced domain that included a single stack plate and sections of resonance tube excited with an oscillating boundary condition (mimicking a mechanical driver) [8]. This model design is illustrated in Fig. 1, which also shows the full device. The group investigated the temperature behavior along the stack plate.

These early modeling results were compared to measurements [9] and predictions from linear acoustic models [10]. Further analytical investigations of the thermoacoustic effect were conducted by Ozoe et al. [11]. Cao's model [8] formed the basis for several later refinements. Further investigation of the temperature behavior, especially deviations from strictly harmonic behavior, was done by Worlikar et al. [12–14]. Besnoin et al. further elaborated on the model by including heat exchangers at either end of the stack, which is comprised of a single plate. This work elaborates on the influence of the placement of said heat exchangers relative to the stack plate [15,16]. The compressible Navier–Stokes equations were implemented into the simulation by Ishikawa et al. [17]. Marx et al. [18–21] combined several previous research efforts and utilized Cao's domain with heat exchangers. The group investigated the non-linearities in the temperature profile, noting that the pressure shows an almost harmonic behavior with respect to time, and the velocity behaves harmonically. The group also identified a critical stack length where non-linearities prevail, which is ≈ 4 times the gas' displacement distance centered around each

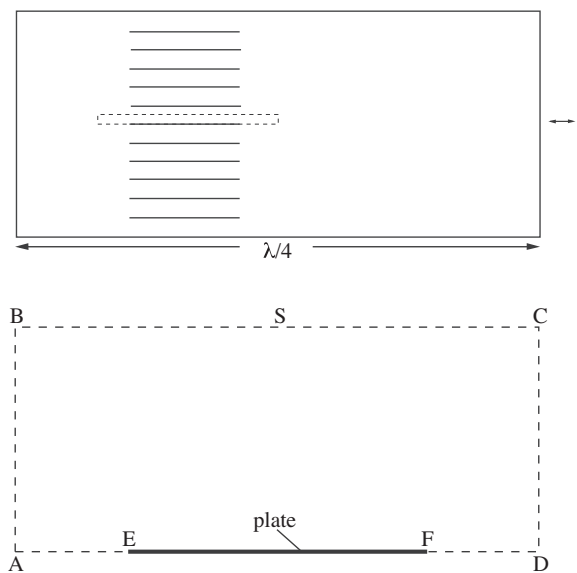


Fig. 1. Computational domain used by Cao et al. [8] for the investigation of the thermoacoustic effect.

stack edge. In addition, the group investigated the influence of stack location on all parameters, including the enthalpy flux. Piccolo et al. utilized the Cao model to investigate the influence of plate spacing and heat exchanger location, as well as the effect of the heat transfer coefficient of the stack on the heat transfer between the solid and gas [22].

2.1. Previous whole device simulations

Zoontjens, finally, transitioned the investigation of the single plate thermoacoustic couple to a computational fluid dynamics (CFD) simulation. Using a commercial code (Fluent), the group developed a mesh that included large sections of resonator and a single plate stack. As with all previous models, the oscillations were introduced by an oscillating pressure boundary condition. This simulation was used extensively to investigate the fluid behavior during the thermoacoustic cycle [23].

Preceding Zoontjens attempts to investigate the thermoacoustic effect using CFD analysis are two additional similar, but simplified, examples. Hanschk et al. used Fluent 4.4.4 to simulate a Rijke tube (which is similar to a classic thermoacoustic engine, except that the oscillations are created with one heated wire screen rather

than a stack) [24,25]. The boundary conditions were a steady velocity inlet and an open end, with a heated screen in the center [24,25]. A further example of the simulation of a Rijke tube was performed by Entezam et al. [26].

Nijeholt et al. [27] used the commercial solver ANSYS CFX to simulate a traveling wave engine inside a Helmholtz resonator, mimicking the design built by Bastyr et al. [28]. Rather than modeling the regenerator geometrically, the group modeled this region with body forces to account for the viscous forces acting on the gas. This includes a volume porosity to account for the solid fraction in the regenerator region. The pressure drop across the heat exchangers and regenerator are accounted for by:

$$\Delta p = \left(C_{fd} + \frac{C_{sf}}{\text{Re}} \right) \frac{\rho u^2}{4d_w} \quad (2)$$

following the work done by Thomas et al. on the friction and heat transfer behavior of gases in Stirling regenerators [29]. This expression depends on the Reynolds number, density ρ , velocity u , and wire diameter d_w , as well as “correlation constants” C_{fd} and C_{sf} which are determined empirically. The group used a very coarse grid, and a time step that used ≈ 100 samples per period of the oscillations (at 53 Hz). Their simulation covered a total time of 1.5 s and did not reach a steady state of constant amplitude [27].

2.2. Present model of a thermoacoustic engine

Building upon this previous work, our group has built a model of a standing wave, $\lambda/4$ resonator thermoacoustic engine. It was modeled after an experimental engine that is open to the environment. Using Gambit, the simulation domain is 150 mm long, 12 mm high. Thirty millimeters away from the left-hand side, the stack channels are located. The stack is 10 mm in length and its channels are 0.5 mm high. The mesh is shown in Fig. 2. It contains $\approx 120,000$ triangular cells. This type of cells is chosen in order to be able to increase the cell density towards the hydrodynamically relevant stack areas. The stack walls were replicated with the mesh, which allows for a more detailed analysis of the flow physics inside the stacks during a thermoacoustic cycle than modeling the stack using porous formulations and body forces. The area to the left of the stacks is referred to as “compliance”. The vertical wall of the compliance is the pressure anti-node and the velocity node.

The horizontal stack walls are defined to have a non-zero thickness and are given a heat transfer coefficient as well as the driving temperature gradient. In order to initialize a the flow field in the computational domain, the left-hand side of the domain is initially defined as a pressure inlet. All walls except the horizontal walls in the regenerator are defined as adiabatic. The heat transfer coefficient on the horizontal regenerator walls resulted in the thermal interaction between the walls and the working gas and the replication of the thermoacoustic effect. To account for turbulence, the $k-\varepsilon$ model is used. This model is chosen because it is the one used most widely and is valid for a wide range of flows [30]. The description of the model refers to the creation (k) and dissipation (ε) of turbulent kinetic energy. Once a steady solution to this condition is found, this area is redefined as a wall and the transient

Table 1

Boundary conditions applied to CFD model.

Parameter	Value
Working gas	Air
Equation of state	Ideal gas
Mean (ambient) pressure	101,325 Pa
Initial pressure disturbance	10 Pa
Pressure outlet	0 Pa gauge (ambient)
Temperature difference (stack)	$T_{hot} - T_{cold} = 700-300$ K
Heat transfer coefficient	50 W/m ² K
Turbulence model	$k-\varepsilon$
Transient time step	10 μ s

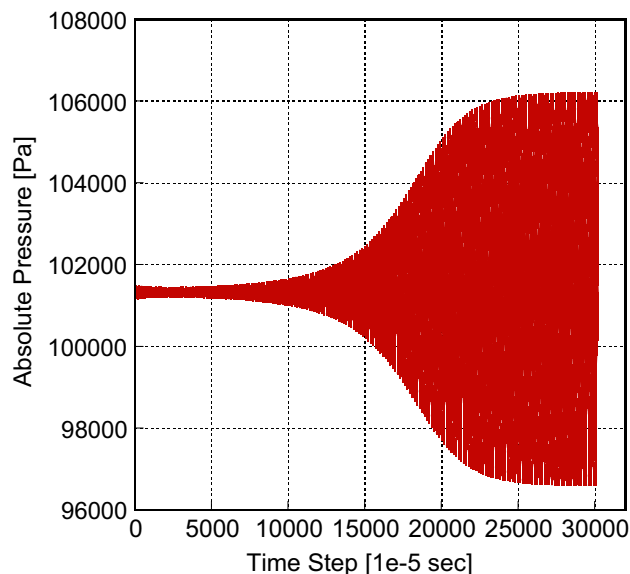


Fig. 3. Simulation of pressure oscillations due to the thermoacoustic effect a CFD analysis as developed by Zink et al. [31].

simulation is started. All other boundary conditions are left unchanged. During the transient simulation, the temperature gradient is the only driving force in the system and is the sole cause of the amplification of the pressure amplitude. The absolute pressure at the compliance wall is used as the metric to illustrate the onset of strong thermoacoustic oscillations. The applied boundary conditions are summarized in Table 1.

Fig. 3 shows the achieved oscillations through the analysis of pressure at the pressure anti-node. Our model is an advancement of previous simulations of thermoacoustic engines because it used a physical working gas and a heat input as its sole driving force (rather than imposed oscillations) [31]. The oscillation frequency is 614 Hz, so the transient time step yields a resolution of 164 time steps per period. Since the operating frequency is determined by the overall length of the device this does not change when the cooling stack is added to the model.

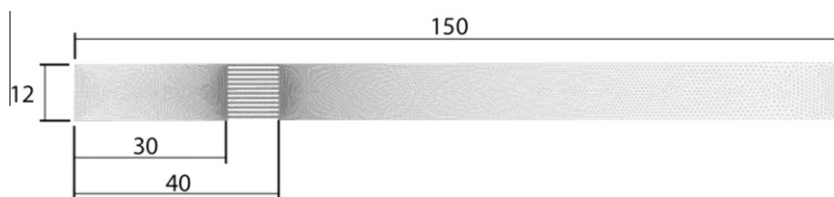


Fig. 2. Simulation domain used for the initial simulation of the thermoacoustic effect by Zink et al. [31].

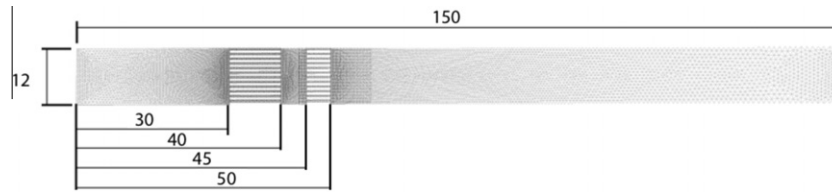


Fig. 4. Simulation domain used for cooling, TAE with a secondary cooling stack.

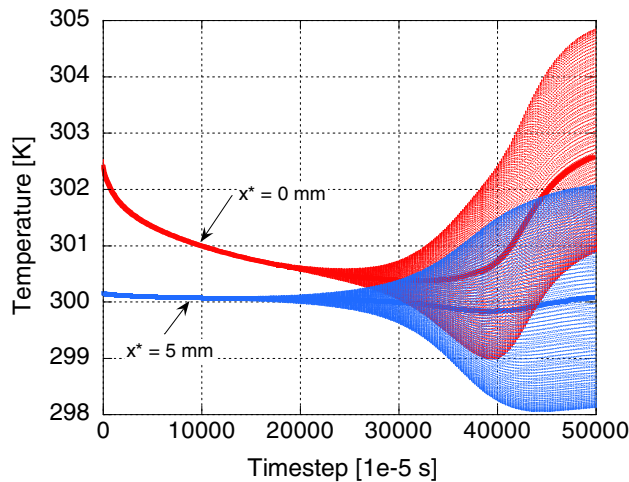


Fig. 5. Temperature behavior across the cooling stack.

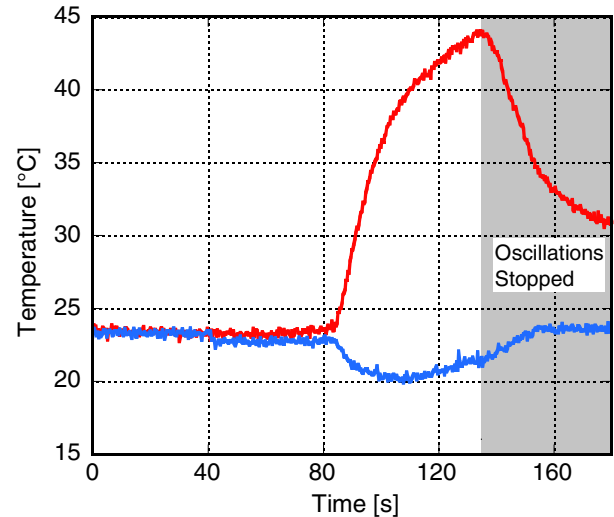


Fig. 6. Experimental demonstration of heat driven cooling.

3. Simulation of thermally driven cooling

As a continuation of our simulation efforts, we developed a model of a thermoacoustic refrigerator. Thermally driven cooling requires a secondary stack where the pressure amplitude of the sound generated in the driving stack can be attenuated and withdraw thermal energy from the gas. Again, this phenomenon formed the basis of the investigations by Cao and ensuing models as described above. In this study, however, this effect is simulated as part of a whole engine rather than a single thermoacoustic couple that is exposed to an oscillating pressure boundary.

3.1. Extended CFD model to include a cooling stack

The present model extends our previous work by the addition of a secondary stack just to the right of the driving stack. As in the driving stack, the cooling stack is modeled with physical flow channels of height 0.5 mm. Its length is 5 mm. The heat transfer coefficient in the cooling stack is chosen to be identical to the driving stack at 50 W/m² K. This model design results in a full representation of a heat driven thermoacoustic refrigerator that does not rely on mechanical excitation through oscillating boundary conditions. Furthermore, our model also advances the investigation of thermoacoustic devices as a whole rather than an individual stack plate. In contrast to the previous models that also simulated thermoacoustic cooling, our model is capable of achieving this by solely applying a heat input to the system. Fig. 4 illustrates the new computational domain. The larger driving stack is located closer to comply the device, the new refrigeration stack is located to its right.

Recalling that the closed end (on the left-hand side) is the pressure anti-node and the velocity node of the standing wave. The thermoacoustic effect requires large pressure variations and non-zero displacement. The cooling stack is located further away from

the closed end. As a result, the pressure amplitude encountered here is decreased, but the velocity amplitude is higher than within the driving stack. These operating conditions are not ideal for the operation of a TAR but they can replicate the concept of thermoacoustically achieved cooling.

3.2. Demonstration of thermally driven cooling

Fig. 5 shows the resulting temperature behavior throughout the cooling stack with respect to time. The variable x^* refers to a local coordinate system with its origin at the left-hand side of the cooling stack. The thin lines show the detailed gas temperature, the thick lines its average value for each period. It is clear that the average temperature achieved in the cooling stack reaches below the ambient temperature of 300 K. Due to the lack of exhausting the pumped heat from the hot side of the cooling stack, the effect only lasts a short period of time.

We compare these results with measurements that were taken from a thermoacoustically driven refrigerator that was set up as shown in the simulation mesh, that is two stacks (utilizing extruded cordierite) located close to each other. The larger stack was heated with an electric heating wire. Due to the low thermal conductivity of cordierite, active cooling of the cold side of the stack is not required. The secondary cooling stack was located close to, but not in contact with the driving stack, towards the open end of the device. Fig. 6 shows the temperature difference across the refrigeration stack. At the onset of oscillations, a temperature difference between the two stack ends is established. This is due to thermoacoustic heat pumping. There was neither a low temperature load nor an exhaust heat exchanger, which explains why the cold side temperature eventually starts rising. The internal heat conduction through the cooling stack's solid areas results in a heat flux from the hot side to the cold side.

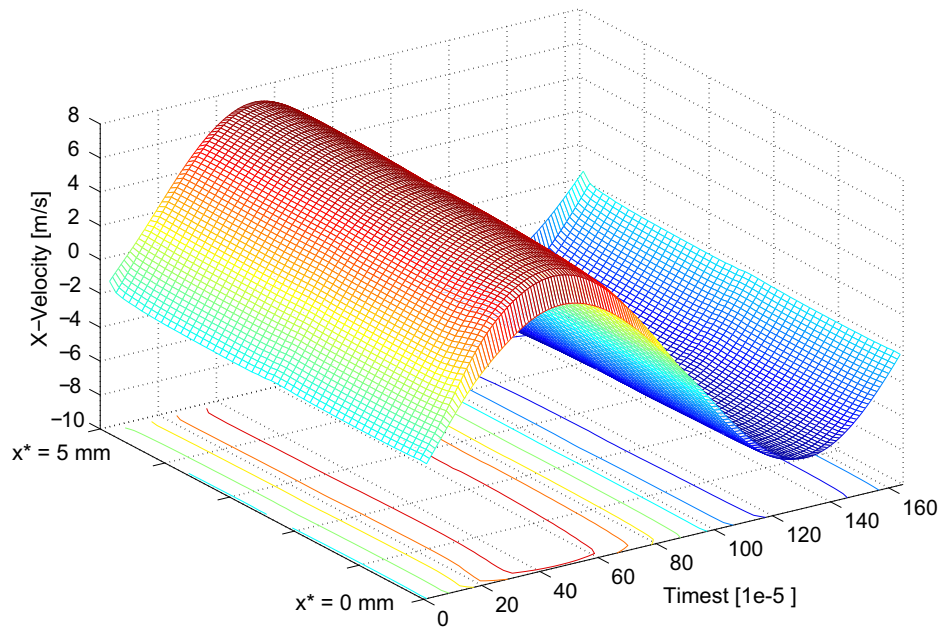


Fig. 7. Velocity distribution throughout the cooling stack, plotted for one entire period.

3.3. Velocity, temperature, and heat flux with respect to time

The flow variables of interest in thermoacoustics are the velocity and temperature, and, as a result, the heat transfer between the solid and gas inside the cooling stack. The following data is extracted from a simulation that reached the state of sustained pressure oscillations, after $\approx 44,000$ time steps. The velocity component is shown in Fig. 7. This figure shows the velocity component of the wave throughout the cooling stack in the center of the channel with respect to x^* and time. Interpreting the velocity behavior, the undisturbed sinusoidal behavior is obvious. Also, the velocity inside the stack is not significantly influenced by the presence of the stack. It must be noted that for the period shown, initially, the axial velocity is negative; that is, gas is streaming from the resonator through the stacks towards the closed end of the device. This supports the explanation of the behavior of the temperature and the heat flux given below.

With the (initially negative) velocity component, the interpretation of the temperature behavior shown in Fig. 8 can be explained. As the velocity becomes positive and continues to increase, there is a disturbance in the temperature contour that propagates through the stack. It dissipates before the opposite end is reached. One half period after this disturbance forms on the opposite end of the cooling stack, a new disturbance starts on the opposite side. This is caused by the reversal of flow and the fluid interacting with the cooling stack. This disturbance is the same phenomenon that was discussed previously by Cao et al. [8] and Marx et al. [20]. The visualization of the temperature with respect to time and axial position is a novel approach to visualizing the effects of the stack on flow variables during the thermoacoustic cycle.

Next, comparing the temperature behavior of the channel center and in the vicinity of the wall, clear difference between the two locations become apparent. The aforementioned disturbance caused by the entrance of the stack starts later and dissipates sooner. Furthermore, the disturbance that develops during the return cycle is more developed than in the center of the channel. Of course, it is the difference between the two temperatures that is proportional to the heat exchange between gas and solid stack wall.

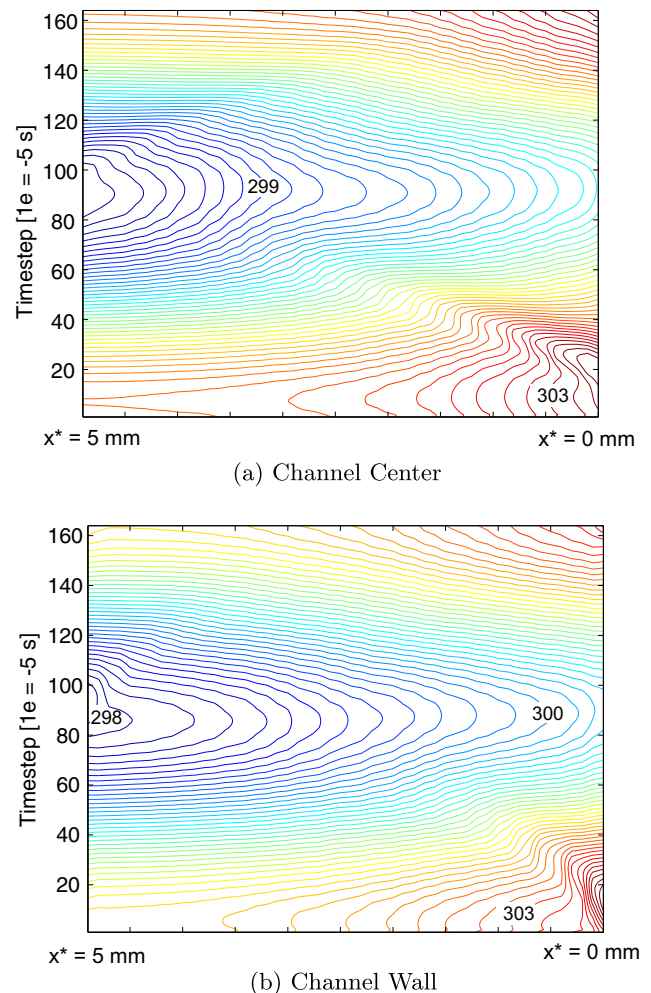


Fig. 8. Gas temperature in the center and on the wall of a channel along the length of the cooling stack, plotted for one entire period.

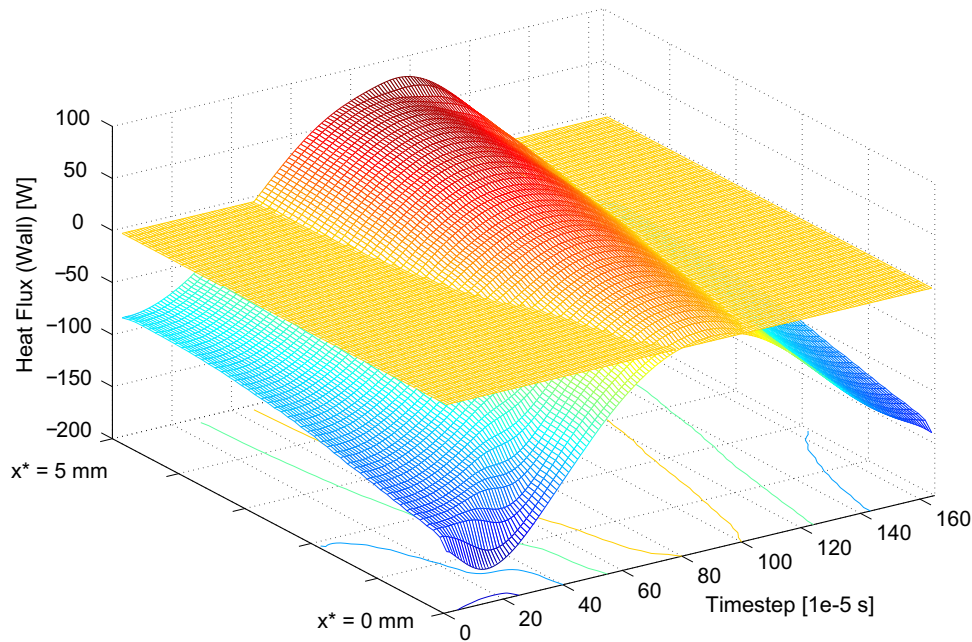


Fig. 9. Heat transfer on the wall Q_w along the length of the cooling stack plotted for one entire period. A $Q = 0$ surface is plotted in addition for improved visualization.

Investigating the heat transfer on the channel wall Q_w throughout the stack, shown in Fig. 9, the sinusoidal behavior with respect to time is apparent. The additional surface cutting through the heat transfer surface illustrates $Q = 0$ for improved visualization of Q_w . The temperature disturbance discussed above affects the heat transfer, which results in a similar disturbance to propagate through the surface plot of the heat transfer. In addition, and more importantly, it is shown that the time averaged heat transfer at $x^* = 5$ mm is less than 0, while the time averaged heat transfer at the opposite end of the stack at $x^* = 0$ mm is greater than 0. A peak is discernible but its visualization is limited due to the discretization of the data extraction during the simulation.

4. Summary and outlook

We have demonstrated the extension of previous modeling efforts of a thermoacoustic engine [31] to include a cooling stack. Rather than previous investigations that illustrated thermoacoustic cooling with oscillations imposed as a boundary condition, our new model only utilizes thermal energy as an input. The oscillations are thus created within the model as part of the simulation. From our previous model, the TAE was driven using a temperature gradient imposed along the horizontal stack surfaces. Oscillations were induced with a small initial pressure disturbance. Once thermoacoustic oscillations were achieved, the temperature of the gas was recorded along the cooling stack. A decrease in temperature below ambient was proven. This is consistent with temperature measurements taken from a similar physical model. Currently, performance is limited by the design of the TAR, specifically the location of the cooling stack in the direct proximity of the driving stack towards the open end of the modeled $\lambda/4$ resonator. This location is responsible for relatively high velocities, and thus viscous losses, while providing a comparatively small pressure amplitude. Locating this cooling stack closer to the pressure node should yield better performance. Nonetheless, the present model serves as an advancement in the demonstration of the principle of thermally driven cooling using CFD analysis.

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